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A METHODOLOGY TO COMPARE REFRIGERANTS BASED ON THEIR THERMOPHYSICAL PROPERTIES

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Abstract:

The comparison of refrigerants in a fair way has been always a matter of discussion. There are some properties like isentropic COP, volumetric capacities or pressure ratio that can be compared easily and supply some relevant information. However, how they are translated into real systems is not so direct. For instance, isentropic COP is obtained under the assumption that all the refrigerants are able to work in the same condensing and evaporating temperatures, but these variables depend on the temperature lift of the secondary fluid, and, in real systems, they will not be the same. The size of the heat exchangers and hence, the cost, depend on heat transfer coefficients and pressure drop, and extrapolating this data in a straightforward way is not easy.

In this work, the authors propose a methodology to compare different refrigerants based on the external operating conditions. The approach compares the system performance not only in terms of efficiency parameters but also in term of operation map, size, cost of the system and refrigerant charge. The developed methodology can be a guide that researchers and students can use to estimate qualitatively the differences among refrigerants.

Keywords: Refrigerants, Efficiency, Heat Tranfer, Thermophysical properties.

1. INTRODUCTION

Comparing different refrigerants for a defined application is not an easy question, and in order to simplify this kind of evaluation, it is quite common to find in the literature approaches that simplify too much the problem, losing important information [1], [2].

In that perspective, for instance, a quite common comparison is based on assuming the same evaporating and condensing temperature for the different refrigerants, and this is something that considering a system working between heat sinks, and sources of finite capacity are not real.

Based on that, this work illustrates a methodology that maintains the simplicity of the calculations (they can be done with an Excel sheet), but allows a systematical comparison among different refrigerants in such a way that aspects like efficiency, cost of the system, operation range or required charge can be assessed at least qualitatively.

2. THEORETICAL BACKGROUND

The comparison among the fluids is done in two parts. In the first one a performance analysis is done, and in the second one an analysis in terms of cost of the equipment is presented.

Performance analysis

- Thermophysical evaluation of the refrigerant properties.
- Ideal COP in the predefined conditions.

Equipment dimensioning

- Required evaporation area/ heat transfer coefficient.
- Required condensation area/heat transfer coefficient.
- Volumetric refrigerant effect.

The calculation of these relative values has been done using the expression:

$$R_V = \frac{V_{refrigerant} - V_{R290}}{V_{R290}} * 100 \qquad (1)$$

Where V means the variable under analysis. The reference refrigerant used for this explanation is propane (R290). Other refrigerants may be used as reference.

2.1 Performance Comparison

In this section, the refrigerant behaviour given some boundary conditions is studied. The boundary conditions are determined by the secondary fluid temperature difference in the evaporator and in the condenser. In order to fix the required cycle parameters, the superheat was also fixed. Based on that, the evaporating and condensing temperature were determined in order to maximize the theoretical COP of the system.

Condensation temperature calculation

For the calculation of the condensation temperature, a methodology based on the pinch point between refrigerant and secondary fluid has been followed. Usually, in order to calculate the COP of the system, it is assumed that the temperature of the secondary fluids do not change (thermal reservoirs) and Carnot cycle is used in order to approach the maximum COP that could be reached by a refrigerant.

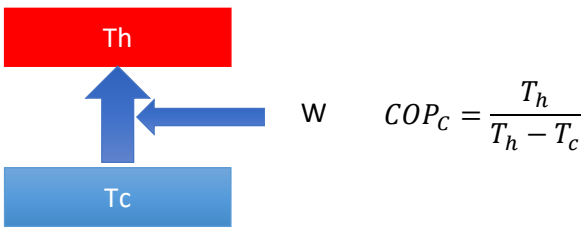
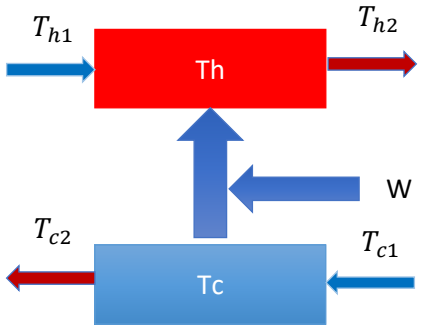


Figure 1: Theoretical Background-Ideal cycle.

This assumption implies that all the refrigerants can have the same condensing and evaporating temperature.

In the real world, a heat pump is a system designed to heat/44cool a fluid therefore the approach of considering that fluid as a thermal reservoir is not the most suitable approach and also have the limitation that some relevant parameters like the discharge temperature in order to heat water to a given temperature cannot be calculated properly. In order to describe the ideal cycle in these situations, Lorentz cycle must be considered.



$$COP_L = \frac{\overline{T_h}}{\overline{T_h} - \overline{T_c}} \qquad \bar{T} = \frac{T_1 - T_2}{ln\frac{T_1}{T_2}}$$

Figure 2: Condensation - Pinch approach

Based on that, figure 3 shows the typical temperature map of the refrigerant and the secondary fluid that can be found in a condenser of infinite area.

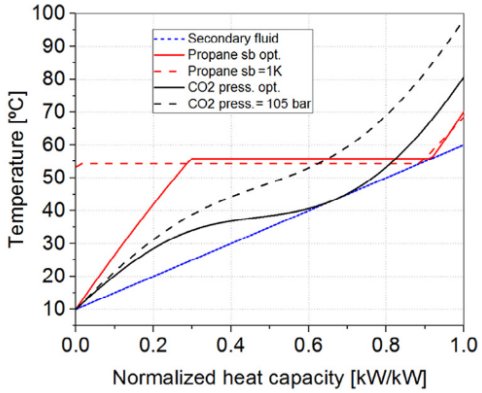


Figure 3: Temperature evolution across the condenser for R290 and CO2 for a heat exchanger of infinite area

Figure 3 shows that for a heat exchanger of infinite area, there are two pinch points in the heat exchanger, therefore from the theoretical point of view, this will determine the optimum condensing temperature and the optimum subcooling for a refrigerant working in a condenser and heating up a fluid from a temperature T1 to a temperature T2. Therefore, the condensation temperature is a refrigerant dependent variable and cannot be fixed. For instance, in order to heat up water from 50ºC to 60ºC just by it thermophysical properties, R290 has an optimum condensing temperature of 59ºC meanwhile R32 has it at 57ºC.

Evaporation temperature calculation

The temperature profile in the evaporator is simpler as it only shows a pinch point. Therefore, the definition of the optimum evaporating temperature as a function of secondary fluid can be done taking into account the secondary fluid inlet and outlet temperatures and the superheat, see figure 4:

$$T_{c1} - T_{c2} > sh \rightarrow T_{evap} = T_{c2}$$

$$T_{c1} - T_{c2} < sh \rightarrow T_{evap} = T_{c1} - sh$$

Figure 4: Evaporation - Pinch approach.

The optimum cycle is defined for a given $Dt_{secondary\ fluid}$ in the condenser and the evaporator and a given superheat, subcooling, condensing and evaporating temperatures calculated according to previous paragraphs, and an isentropic compression.

Basic refrigerant performance parameters

Based on the methodology presented previously to define the ideal cycle, the condensation/evaporation temperatures, the performance of the refrigerants and many relevant parameters of the cycle have been calculated. These parameters are:

- Isentropic discharge temperature.
- Density.
- Pressure ratio.

$$P_r = \frac{P_{cond}}{P_{evap}} \quad (2)$$

- Isentropic COP.

$$COP_{isen} = \frac{h_{2isen} - h_1}{h_2 - h_1} \quad (3)$$

2.2 Equipment size

Minimum required heat exchangers area

To establish a fair comparison of the performance of different refrigerants in refrigeration systems in a systematic way could be quite challenging as it requires the right design of the heat exchanger for each refrigerant. Different refrigerants have different heat transfer coefficients, different pressure drops, different mass flows for a given capacity and the heat exchanger must be designed attending all these particularities.
In order to establish an approximate estimation of the behaviour of each refrigerant from the point of view of heat exchangers that allows a fair comparison among them a simplified approach has been developed.
The approach is developed assuming that the heat exchange is produced in a horizontal pipe.
The model uses the Moreno-Quiben and Thome correlation for pressure drop (PD) [3] [4] and the Wojtan and Thome correlation for heat transfer coefficient (HTC) [5] [6]. These correlations have been chosen because they are supported by the literature, they are based on the flow patterns that have the refrigerant along the exchanger, and the properties are calculated according to the pattern. Figure 5 shows an example of the correlation results, where for each quality, there is a different value of HTC and PD, due to phase change and two-phase flow patterns.

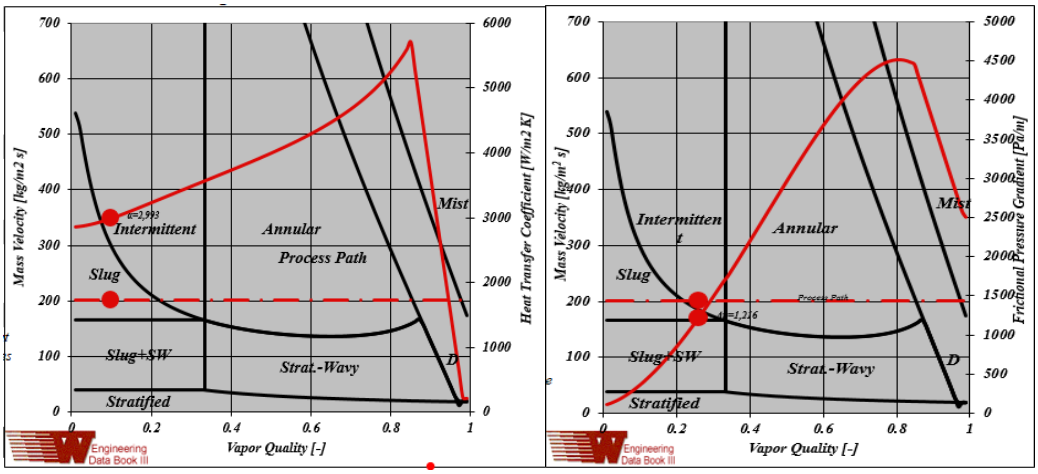


Figure 5: Correlations representation for R290 in evaporation. Heat transfer coefficient (left) and frictional pressure drop (right), Wolverine Engineering Data Book III.

The main idea of this methodology is to find for each refrigerant, the combination of number of circuits [7] and number of tubes (refrigerant-side area) that meets the restrictions imposed and obtains the minimum size of area necessary.
The restrictions for the calculation of the evaporator are:

- The number of circuits has to be a whole number.
- The gas velocity must be between: 8 m/s < Gas velocity < 15 m/s.
- All cases have to reach a temperature difference between inlet and outlet of 5 ºC.
- The total pressure drop on the evaporator has to be lower than 0.5 bar.

For the condenser:

- The number of circuits has to be a whole number.
- All cases have to reach a temperature difference between inlet and outlet of 5 ºC.
- The total pressure drop on the evaporator has to be lower than 0.5 bar.

Once all the restricted conditions are met, the equipment is sized, and thus the total heat exchanger area and the heat transfer coefficient are obtained. The comparison between the results can be considered as valid since each refrigerant is being looked for its optimal configuration in the same operating conditions. In this way, and by not setting previous sizing parameters, each refrigerant will find the most advantageous solution to satisfy the required capacity.

Refrigerant charge estimation

To estimate the refrigerant charge, the formula below proposed by Steiner [8] has been used. It is important to comment that this is also an ideal estimation, and it is intended to supply an orientative value that could be useful to compare the results between refrigerants. Hence, data are presented as a comparison with R290.

$$\varepsilon = \frac{x}{\rho_G}\left[\left(1+0.12(1-x)\right)\left(\frac{x}{\rho_G}+\frac{1-x}{\rho_L}\right)+\frac{1.18(1-x)[g\sigma(\rho_L-\rho_G)]^{0.25}}{\dot{m}_{total}^{2}\rho_L^{0.5}}\right] \quad (4)$$

ε: void fraction.
X: Vapor quality.
ρ_G: Gas density.
ρ_L: Liquid density.
g: gravity acceleration.
σ: surface tension.

Volumetric refrigerant effect

This parameter represents the refrigerant's volume that has to be displace in order to achieve the desired capacity on the condenser. This value is used to compare the required compressor volume needed for each working fluid. $\Delta h\ condenser$ represents the enthalpy difference in the condenser.
Volume required to reach the required capacity:

$$Volumetric\ flow\ rate\left(\frac{m^3}{s}\right) = \frac{Heating\ capacity\ (W)}{compressor\ inlet\ density\left(\frac{m^3}{kg*s}\right)\Delta h\ condenser\left(\frac{kJ}{kg}\right)} \quad (5)$$

3. RESULTS

This methodology has been implemented in an excel sheet with the available macros from coolprop. Figure 6 represents a comparison among different fluids obtained using this methodology. This comparison should be taken as an example of a possible output from a calculation using the developed methodology.

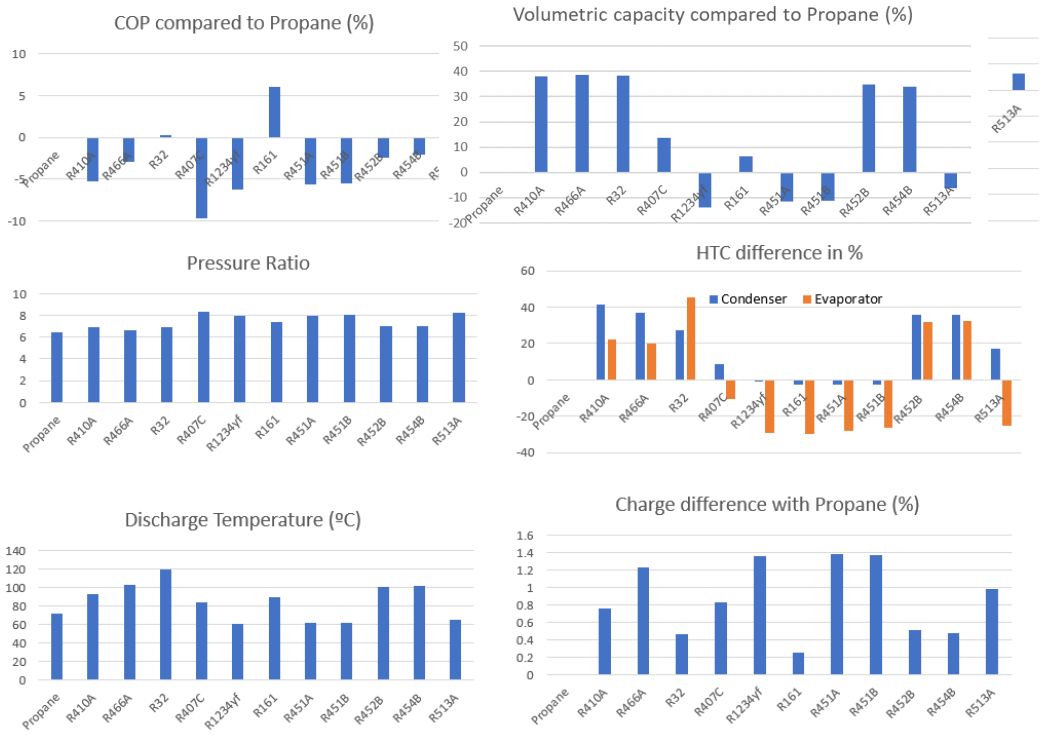


Figure 6: Comparative results among different refrigerants obtained with the used methodology.

4. CONCLUSIONS

This paper has shown a methodology which allows a qualitative and easy comparison among different refrigerant fluids. This comparison does not only allow to compare them in terms of performance but also in terms of size and cost, considering it directly related to the weight and size of the components, of the different components and allows to give indications about a preliminary sizing of heat exchangers, required charge and operation map.

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